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ABSTRACT

Performance of a reciprocating refrigerator compressor with mixed-refrigerants (MR) was investigated experimentally. The current, standard methods of testing compressors and presenting performance data were found unsuitable for MR, and a new experimental apparatus was designed for a direct flowrate measurement with MR. Representative test data for Refrigerant R-12 and for a mixture of R-22/R-114 were presented and analysed. It was shown that by excluding the suction-gas heating effect through a direct measurement of refrigerant temperature prior to entering the compression chamber, data for various refrigerants could be correlated in a consistent thermodynamic framework. Results show a direct relationship between the overall, mechanical work of compression and the theoretical work for a reversible, isentropic compression. Furthermore, volumetric flowrates were well-correlated by a simple "constant volume re-expansion" model for a wide variety of refrigerants and operating conditions. Finally, a simple model was outlined for predicting flowrate and power consumption for the compressor under investigation for any refrigerant with either single or multiple components.

PERFORMANCE OF RECIPROCATING COMPRESSORS WITH NON-AZEOTROPIC MIXTURES

In the recent years there has been a growing interest in the use of mixtures of

Refrigerants (MR) in vapor compression devices. Most of the current work on MR application can be divided into two major categories: one aimed at achieving a higher cycle efficiency as a result of non-isothermal phase-change with mixtures [1], and the other aimed at improving the overall system performance achievable through capacity-modulation by changing MR compositions [2]. Regardless of which of these two applications is of interest, it is essential to understand the performance of a device operating with MR and to be able to compare it with a similar device operating with a single component-refrigerant.

Before one can compare the overall device performance, however, it is necessary to measure and understand the impact of MR on each of the individual components. It is only after the component performances have been well characterized that one can study their various interactions that affect the overall system performance. Thermodynamic implications of the use of MR on heat-exchanger performances (condensers, evaporator, and suction-line) have been presented elsewhere [3]. It is the objective of this paper to present and interpret new data on the performance of a reciprocating compressor with MR. The major goal of the research project was to provide answers to the following questions:

1. How does a reciprocating refrigerator compressor perform with MR?

2. Is there a uniform thermodynamic framework in which performance data with various MR and single component refrigerants can be uniformly correlated?

3. Can a simple model predict the compressor performance (flowrate, power, etc.) that is of interest to the refrigeration system designer regardless of the choice of single-component or mixed refrigerants?

COMPRESSOR CALORIMETRY

The current practice in the refrigeration industry is to measure the performance of a compressor in a bench calorimeter (usually a secondary refrigerant calorimeter) at a pre-defined set of 'standard' conditions. When the performance over a broad range of operating conditions is of interest, results are often presented as plots of compressor watts and evaporator cooling capacity as a function of evaporator saturation temperature, with condenser saturation temperature as a parameter. Unfortunately, such plots are applicable only to that particular single-component refrigerant and can not be translated readily to a different choice of working fluid. Another drawback of the standard calorimetry is that these performance curves obtained by 'in vitro' measurements often do not accurately represent the 'in-vivo' performance of the same compressor in an actual refrigeration unit. The main reason for such discrepancies is that the compressor cooling environment is not the same for the two cases.

The shortcomings of the standard calorimetric procedures become even more evident when one considers compressor performance with MR. The most critical problem being that for non-azeotropic mixtures there is no unique saturation temperature and any direct comparison with 'standard' curves described above becomes meaningless. Furthermore, from an experimental point of view, it is difficult to maintain a constant composition of the circulating mixture in the standard calorimeter equipment over a broad range of operating conditions. In order to obviate these difficulties a new experimental procedure was developed for measuring compressor performance with MR. A schematic of the apparatus is shown in Figure 1. The compressed refrigerant is condensed in a coaxial watercooled condenser with a temperature controller to maintain a constant temperature subcooled liquid at the inlet to the adjustable expansion valve. The evaporator is electrically heated with another temperature controller to maintain a constant suction gas temperature at the inlet of the compressor. The basic approach is to use a direct mass flow measurement using a vibrating U-tube flowmeter prior to the expansion valve, rather than relying on an 'indirect' calorimeter measurement based on the evaporator energy input. A turbine flowmeter for measuring volumetric flow in the suction-line was also tested during the course of the project but it did not yield as reliable and accurate an estimate of the mass flowrate as the mass flowmeter, perhaps due to the interference caused by the circulating oil. In addition to the standard measurements of pressures at the compressor suction/discharge and temperatures at various points in the cycle, a specially installed thermocouple at the piston slot intake inside the compressor was used to obtain the refrigerant temperature as close to the actual cylinder intake as practical. The reason for measuring this cylinder inlet temperature and the key role it plays in correlating compressor performance will be discussed later in this paper.

A series of tests were performed over a wide range of suction and discharge pressures using refrigerant R-12 as well as different compositions of a binary mixture of refrigerant R-22 and R-114 (dichlorotetrafluoroethane). Composition of the circulating mixture was measured in a gas-chromatograph by taking a sample after reaching steady-state operation. The mixture composition was found not to vary significantly over a range of operating conditions, mainly due to the very small charge hold-up in the system. A small sample of typical test results is shown in Table I for refrigerant R12 and for a mixture of R22/R114 with a composition of 37% by mole (23% by weight) of R22.

THERMODYNAMIC MODEL OF THE COMPRESSOR

Before presenting analyses and plots of test data, it is important to discuss the general thermodynamic viewpoint of the overall compressor performance that has allowed us to successfully correlate data with different refrigerants. The compression process is assumed to consist of three separate steps:

1. Suction-Gas Heating

The refrigerant vapor entering the compressor at a suction temperature T_s is heated up to a temperature T_{cyl} before entering the compression chamber. It is generally well accepted that this heating effect is of major importance to the overall performance of the compressor [3]. It should be noted that the magnitude of this heating is dependent not only on the compressor design but also on the cabinet application since the latter influences the cooling environment around the compressor. Recognizing this as the major factor in discrepancies between "in-vitro" and "in-vivo" performance of a compressor, a special effort was made to measure T_{cyl} by an internal thermocouple in this study. In retrospect, without such an approach it would not have been possible to correlate the data with various working fluids.

2. Compression inside the Cylinder

Having accounted for the suction-gas heating, it is assumed that the actual mechanical work of compression is related to the "thermodynamic difficulty" of the compression task as represented by the work required for a reversible, isentropic compression starting at P_s & T_{cyl} and ending at P_d . Since by considering only a ratio of reversible work to the actual mechanical work all of the details of the compression process (e.g., valve pressure drops, wall heat transfer, dynamics in the discharge tube, etc.) are lumped into a single efficiency factor, such an assumption may rightly be criticized as a gross over-simplification. Fortunately, the results based on such an approximation appear to be sufficiently well-correlated to warrant its use for system analyses and design.

The volumetric flowrate at the cylinder inlet conditions (P_s , T_{cyl}) is assumed to be correlated by a "constant volume re-expansion" model, which stipulates that a small volume of trapped refrigerant re-expands from the density at the end of the compression back to the suction density, thereby reducing the net volumetric flowrate to below the theoretical maximum value. The specific volume at the end of compression (v_d) can be calculated from the discharge pressure P_d and the enthalpy at the end of compression, the latter being equal to the sum of the cylinder inlet enthalpy and the specific mechanical compression work.

3. Conversion from Electrical to Mechanical Power

For the purpose of this work it was assumed that the efficiency of conversion of electrical power to mechanical (shaft) power is identical to that given by the standard motor parformance curves, which are assumed to be available for the motor in question. Consequently, it is a simple matter to obtain the motor efficiency (E_{mot}) either from the measured electrical watts for data analyses, or from the mechanical power requirements for model predictions.

CORRELATION OF RESULTS AND DISCUSSION

From available measurements, the overall mechanical compression efficiency (E_{mech}) is calculated as follows:

1. From P_s, T_{cyl}, P_d, and m_{ref} mechanical power for reversible, isentropic compression is calculated using the refrigerant properties (see [4] for MR thermodynamic properties calculation procedure).

2. Actual mechanical (shaft) power is calculated from the measured watts and the motor efficiency

3. E_{mech} is the ratio of the reversible power to the actual mechanical power

As seen from Table I, despite the fact that the operating conditions vary over a wide range andthat the vapor pressure for the mixture is considerably lower than that for R-12, E_{mech} varies over a very narrow range of 66% to 69%. This is quite an amazing result considering the fact that the simple thermodynamic framework ignores almost all the subtleties of the compressor component behavior. For the purpose of model predictions, a constant value of 67.5% should provide adequate accuracy.

The second part of data analyses involves a comparison of volumetric flowrates, which are calculated as follows:

1. From p_s, T_{cyl}, and m_{ref} the specific volume at the cylinder suction (v_s) and the total volumetric flowrate is calculated.

2. The specific volume ratio is obtained from v_s and the specific volume at the end of compression v_d.

The validity of the "constant volume re-expansion" model can now be tested by a plot of volumetric flowrate against the specific volume ratio v_s/v_d as shown in Figure 2. It should be noted that many more test data are plotted in the Figure than the small sample shown in Table I. Furthermore, several test data for a different mixture composition are also included. Once again it can be observed that despite the drastic nature of model assumptions, the linear correlation is remarkably good over a considerable range of operating conditions. However, two important points should be noted:

1. The linear correlation does not predict a volumetric efficiency of 100% at a specific volume ratio of unity. This is most likely a result of not accounting explicitly for suction and discharge pressure drops, effects that become increasingly important when suction and discharge pressures are equal.

2. The "effective" trapped volume calculated from the slope is not the same as the geometric clearance volume of the compressor. Qualitatively it can be argued that the combined effects of

valve dynamics and refrigerant back-flow are responsible for the steeper drop in the volumetric flowrate than that based on the geometric clearance volume alone.

CONCLUSION

In conclusion the following answers may be offered to the three questions posed in the beginning of this paper:

1. There appears to be no fundamental difference in the performance of a reciprocating compressor with mixed-refrigerants from that with a single component refrigerant as long as a proper framework is used to compare the two .

2. A general thermodynamic framework was described for correlating the performance of the compressor with any choice of refrigerant. An important conclusion was that one needs to exclude the suction-gas heating effect from such generalised correlations.

3. Based on the success of the thermodynamic approach to data analyses, a simple but accurate model was proposed for predicting flowrate and power consumption for the compressor operated with any choice of working fluid as long as its thermodynamic properties were available.

Finally, it is hoped that this work will prove useful not only to those interested in MR applications, but also to many others interested in a compact representation of compressor jerformance curves and in reconciling "in-vitro" compressor tests with "in-vivo" performance in a refrigerator.

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TABLE I: TYPICAL TEST RESULTS AND ANALYSES

	Single Component R-12				Mixed Refrigerant 37% by Mole R22 63% by Mole R114		
Experimental Data							
Suction Pressure P_s(psia)	15.7	19.0	20.9	28.2	12.3	14.3	16.0
Discharge Pressure P_d(psia)	162.2	164.3	171.2	170.5	126	125.3	125.5
Suction Temp T_s(F)	88	44.7	88.5	90.6	90.2	86.7	89.2
Cylinder Inlet T_{cyl}(F)	201.8	179.8	196.6	192	184.3	179.4	177.7
Mass Flow M_{ref}(#/h)	14.4	18.7	20.8	29.1	13.2	16.2	18.7
Electrical Power (Watts)	242	276	303	366	195.9	215	230.5
Data Analyses							
Emotor(%)	71	72	72	72	67	68.5	70
Emech(%)	68	68	69	67	67	66	67
Suction Volume Flow (cuft/h)	54.8	55.7	57.7	58.5	52.4	55.0	56.3
Specific Volume Ratio (v_s/v_d)	7.78	6.64	6.31	4.81	8.1	7.22	6.52

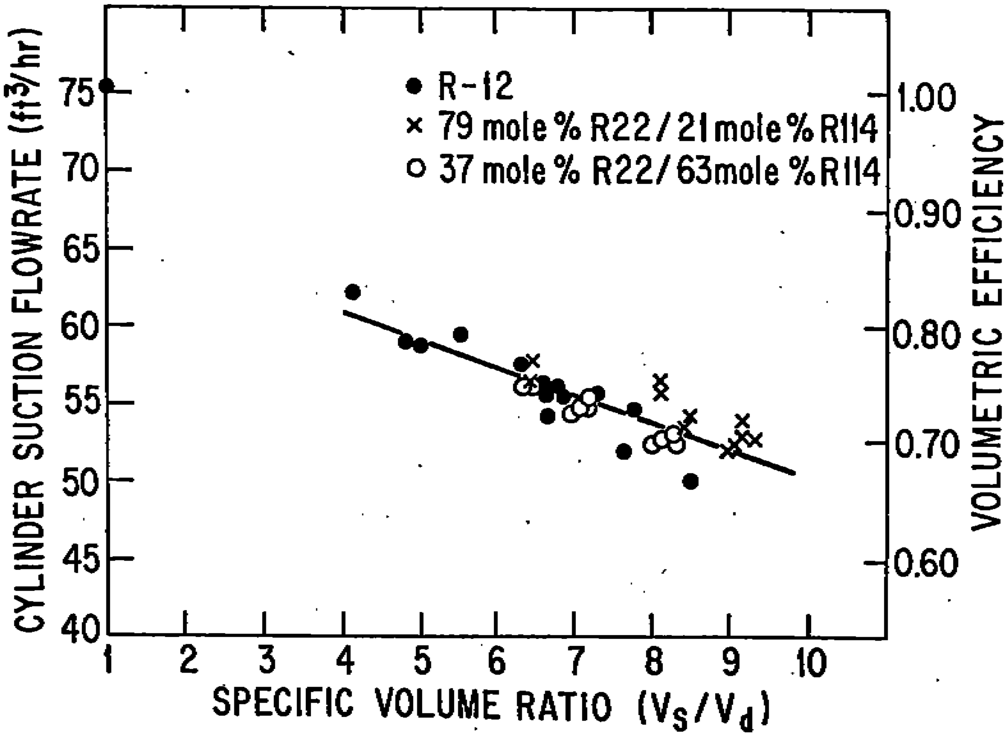

Figure 2: <u>A Comparison of Volumetric Flowrates</u>

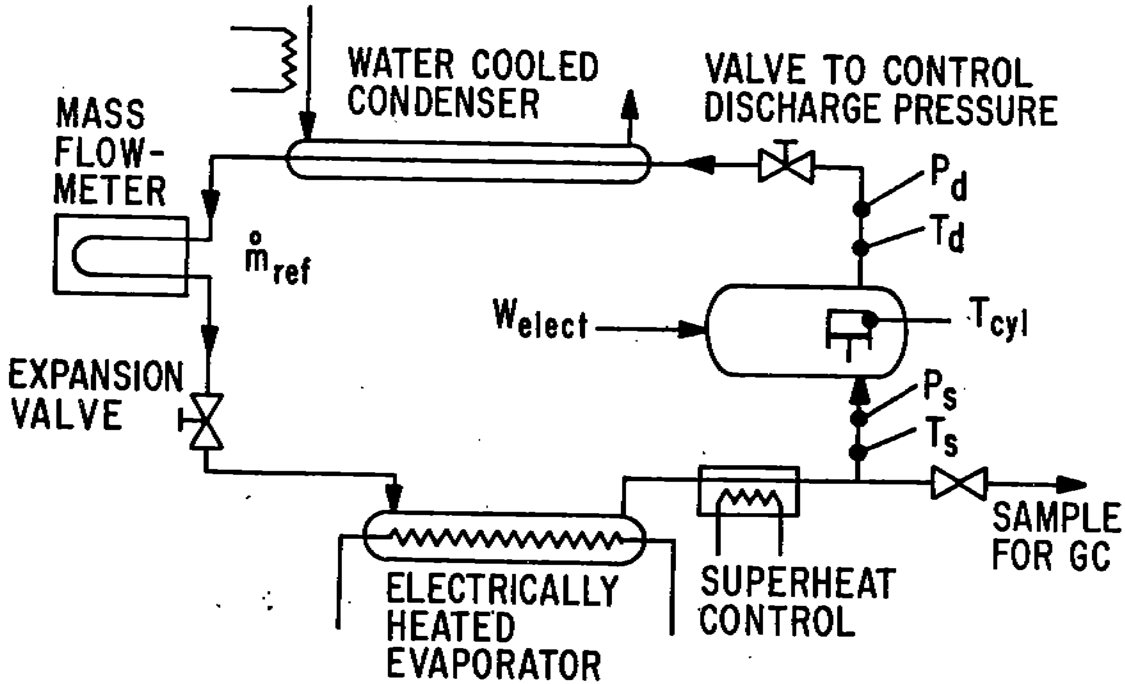

Figure 1: Schematic of the Experimental Apparatus